CHAIN DRIVE WITH PIVOT STEEL CHAINS OR ROUND STEEL CHAINS AS TRACTION MECHANISM FOR SPUR GEARS DRIVING A POLYGONAL CHAIN SPROCKET

The invention relates generally to a chain drive having spur gears with a polygonal chain wheel for pivot steel chains or round steel chains and more particularly, to an arrangement that reduces variations in velocity and acceleration of the chains.

More specifically, the invention relates to those drive arrangements comprising at least a gear wheel attached to the chain sprocket axis with the chain sprocket axis being rotatively connected to a driven gear wheel having a varying size of the pitch circle.

BACKGROUND

Chain drives are generally used in material handling and drive technology for lifting applications and also for continuous conveyors. The compensation of the polygonal effect has been tried with different, mostly complicated compensating gears.

Practical applications to reduce the polygonal effect are hardly known due to the expensive design of compensating gears.

The chain drive designated before is known from the publication DE 15 31 307 A1 (counterpart UK publication 1,167,907). In this publication a gear wheel is driven with a varying pitch circle diameter, where a minimum radius coincides with the center of a chain pocket, while the largest radius coincides with a point at which the chain runs along the pitch circle diameter of the chain wheel.

However, with this embodiment an optimal compensation of the variations of velocities and acceleration is not possible, because the equivalent polygon needs an additional reduction of the pitch circle in the case of round steel chains at the position of the tooth middle of the chain wheel. Furthermore the known proposition does not consider that the gear tooth cutting at the concave points of intersection of the discontinuous rolling curves can not be manufactured with noncircular gear forming methods in a technically and economically feasible manner.

Also the geometric shape of the rolling curve between the points of intersection remains undefined for the individual sections. The radial und tangential velocity variations of chain drives are designated as polygon effect and constitute a problem which is well known and has been investigated many times. The chain running around the driving chain wheel results in undesirable variations of velocities and accelerations in radial and tangential directions.

SUMMARY OF THE INVENTION

The invention aims at compensating the tangential accelerations and to prevent undesired vibrations of the chain drive. It is an object of the proposed invention to solve this problem in such way, that the driven gear wheel and the driving gear wheel consist of noncircular gear wheels having a gear ratio adjustment and a positional arrangement that the smallest angular velocity coincides with the corner middle of the chain sprocket polygon and the greatest velocities occur at the middle of the chain sprocket polygon long straight lines.

This proposition consists advantageously of one or several gear sets with variable angular velocity. In doing so, the rolling curves of the gear wheel sets are shaped in such a way that they consist of continuous toothed sections of the rolling curves of noncircular gear wheels and have such a position relative to the chain wheel that the tangential variations of the chain velocity is avoided.

The noncircular gear mesh transforms a constant drive angular velocity into a variable driven angular velocity in such manner, that during an increasing or decreasing distance of the chain to the center of rotation an opposite decreasing or increasing angular velocity is created and thereby the desired tangential variation of the velocity is achieved.

However, arbitrary gear ratios with one or several noncircular gear sets cannot be realized. Only certain average gear ratios are feasible. The design results in a special advantage to use this approach for both pivot chains or roller chains (hereafter referred to as pivot chains) with equal angular sections and round steel chains or round link chains (hereafter referred to as round link chains) with unequal angular sections of the equivalent polygon of the chain wheel. Round link

chains are not limited to chains with circular cross-sections but also include elliptical and other rounded chain link cross-sections.

In case of round link chains small chain wheels with a small number of teeth are also designated as chain pinions or sprockets. For the purpose of the invention the number of teeth of the sprocket being even or odd is meaningless. In case of round link chains it is advantageous that the driven gear wheel exhibits a larger radius of the pitch curve at the middle of the shorter straight line than the radius at the middle of the longer straight line and both shorter and longer straight line sections form the equivalent polygon.

A further design option is facilitated through a spur gear with one or several noncircular gear meshes, where at least the last gear mesh is embodied as noncircular gearing. The gear ratio and other parameters can be influenced by one or more such noncircular gear mesh.

According to further aspects the velocities and accelerations of driven sprockets with pivot chains can be influenced by a design which exhibits the same number of continuous rolling curve sections as the number of teeth of the sprocket.

The advantage is an almost perfect motion producing an equal chain velocity at each angular position of the sprocket.

According to another aspect of the invention related to round link chains, the driven gear wheel at the pitch curve has a number of continuous rolling curve sections that are twice the number of teeth of the sprockets. Thus, the desired motion also occurs for round link chains.

The continuous rolling curve motion is also accomplished by a driving gear wheel with continuous rolling curve sections at the pitch curve circumference. According to other features of the invention there is provided an arbitrary number of continuous rolling curve sections equal to or more than one in number for driving gear wheels for pivot chains. Thus, the gear ratio can be accordingly adjusted.

An analogous application for other chain types is achieved by providing an even number of continuous rolling curve sections for driving gear wheels for round link chains.

According to this aspect of the invention, the choice of the gear ratio of the driving gear wheel to the driven gear wheel is adjusted by the number of continuous rolling curve sections of the driving gear wheel and its related pitch angle.

According to another aspect of the invention, the geometric shape of the continuous rolling curve sections is designed in such manner that the constant driving angular velocity results from multiplying the driven angular velocity ($\omega_2 = \omega_1 / i$) by the gear ratio at the polygon corner mid-point, i_m, and the cosine of the driven angle (φ_2) to achieve $i = i_m \cos\varphi_2$. Appropriate rolling curve shapes satisfying this relationship can be used.

A further aspect of the invention provides continuous rolling curve sections of such geometry that the gear ratio can be approximated by basic or composite polynomials, trigonometric functions, Fourier series or periodic or mathematical approximating functions.

According to the said propositions of the rolling curves it is advantageous to derive the gear ratios at the corner middle of the polygon (i.e., the center of the corners of the equivalent polygon located at radius r_0) define the rolling conditions and

for pivot chains to be subject to

equation (A) $i_m = \varphi_1 / \sin\varphi_{2\max}$ and

for round link chains to be subject to

equation (B) $i_m = \dfrac{\beta_1 + \gamma_1}{\sin\beta_2 + \sin\gamma_2}$.

At the points of intersection between the single rolling curve sections an improvement can be advantageously accomplished in such a manner that the rolling curve sections of the driven gear wheel at the points of intersection exhibit concave, onesidedly bent transition curves with tangential points on the rolling curve sections.

Another development for the transition between the rolling curve section consists of the fact that instead of the tangential transition arcs, double-bent adjustment curves or an undulating curve lies within the tangential points of the continuous rolling curve sections.

Thus, the invention provides that the transition arcs are symmetrical and can be described mathematically at least by a polynomial of fourth order or a modified trigonometric function of at least $x \sin x$.

In practice, fabrication of the tooth gearing of the rolling curve sections can be facilitated in such a manner that the adjustment curves and transition arcs exhibit at the angle of the intersection point with the continuous rolling curve sections a radius of curvature that is equal to or greater than the radius of the manufacturing tool.

A further improvement constitutes a design, where the driven gear wheel is fabricated at least in two pieces separated at the points of intersection, so that the assembly of a primary part and a secondary part results in concave sharp rolling curve intersections without transition arcs or adjustment curves.

Further means to design the transition between two rolling curve sections consist of removing every second rolling curve section and to provide an arc gap with a radial reduction down to a centering radius.

Furthermore it is advantageous to use the sectional gap as both a tool recess and as a centering means for the complementary part. A further embodiment is provided in such manner, that the partial regions with transition arcs or adjustment curves with supposedly non compensative polygonal effects can be compensated by one or additional next higher noncircular driven gear wheels and driving gear wheels by circumferentially correctly positioned arrangements of transition arcs with appropriate gear ratio relative to the centered driven gear wheels and driving gear wheels.

DESCRIPTION OF THE DRAWINGS

The invention may take form in certain parts and in an arrangement of certain parts taken together and in conjunction with the attached drawings which form a part hereof and wherein:

Fig. 1 is a cross section of a chain drive with a noncircular spur gear;

Fig. 2 is a first example of a chain wheel with equivalent polygon and noncircular gear wheel set with equal pitch angles;

Fig. 3 is a second example of a chain sprocket with unequal pitch angles;

Fig. 4 is an illustration of the kinematics of the polygonal effect of the chain B – A running around the chain wheel;

Fig. 5 shows transition curves and adjustment curves between the intersection of adjacent rolling curves at an enlarged scale;

Fig. 6 A is a section through a complementary noncircular gear wheel;

Fig. 6 B is a front view of the complementary noncircular gear wheel;

Fig. 6 C is a front view of the primary part; and,

Fig. 6 D is a front view of the secondary part.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings wherein the showings are for the purpose of illustrating a preferred embodiment only and not for the purpose of limiting the invention, there is shown in Fig. 1 a spur gear (2). Spur gear (2) is shown with a noncircular toothed driven gear wheel (3a) positioned on the chain wheel axis with a traction mechanism embodied by a steel pivot chain (7) or a round link chain (8) and the driven gear wheel (3a) is driven by a noncircular driving gear wheel (4a). The latter is driven by an additional gear mesh embodied by a driving gear wheel (4) and a driven gear wheel (3), which is driven by an electric motor at the drive input side (5). At the drive output side (6) a chain wheel (10) is located on the chain wheel axis (1). At least the last mesh (12) (i.e., 3a) of the spur gear (2) holds a polygonal chain wheel (10). (As used herein, "mesh" means a pair of gear wheels, such as pinion and driven gear wheel, in toothed engagement. Accordingly, last mesh means the last pinion and driven gear wheel in the gear drive train.) It is to be appreciated that the rotational centers of chain wheel 10 and driven noncircular gear wheel (3a) are not only on a common axis (1) (i.e., such as the chain wheel being splined to driven gear) but the angular or circumferential positions of the chain wheel and the driven noncircular gear wheel on the common axis are fixed at set positions to assure the chain wheel polygon corresponds to certain segments of the noncircular driven gear. Similarly, the angular or circumferential position of driving gear 4a is fixed on its axis

to assure meshing of noncircular driving gear teeth with noncircular driven gear teeth.

Means to reduce variations of velocity and accelerations transmitted to the chain wheel (10) consists of a spur gear (2) attached to the chain wheel axis (1) which is embodied by a noncircular driven gear wheel (3a) with a pitch curve of variable diameter rotationally attached to chain wheel axis (1).

"Polygonal" is a term known in the art when used with chain wheels and may or may not refer to the shape of polygonal chain wheel (10). "Polygonal" refers to the shape of straight lines connecting the teeth or pockets on chain wheel (10) and the straight lines of the polygon are a function of whether the chain is a pivot chain (Fig. 2) or a round link chain (Fig. 3). Polygonal chain wheel is used herein in its conventional sense. When the number of pockets or teeth (c) on crank wheel (10) are few in number, other words such as a "sprocket" or "pinion" may be substituted for "chain wheel".

During the chain wheel rotation the polygonal effect is created through the variable lever arm h (φ_2) (See Fig. 4). Generally the longitudinal (chain direction can be horizontal as in Fig. 4, vertical or inclined and "longitudinal" is intended to cover all directions) chain velocity v is calculated from

$$v = h(\varphi_2)\omega_2 \qquad (1)$$

Choosing a law of motion with variable angular velocity

$$\omega_2 = \frac{\omega_1}{i_m}\frac{1}{\cos\varphi_2} \qquad \text{results in} \qquad (2)$$

$$h(\varphi_2) = r_0\cos\varphi_2 \qquad \text{by multiplication}$$

$$v = r_0\frac{\omega_1}{i_m} \qquad (3)$$

as resulting horizontal velocity of the chain independent from the rotational angle φ_2.

Integrating (2)

$$\omega_1 = i_m\omega_2\cos\varphi_2$$

results in the equation defining the angle between driving and driven gear wheel

$$\varphi_1 = i_m\sin\varphi_2 \qquad (4)$$

The desired transmission behavior between φ_1 and φ_2 is now solved with one or more pairs of noncircular gear wheels (3a),(4a) with piecewise continuous rolling curve sections or lobes (9) in such manner, that the partial arc lengths (27) of the driven gear (3a) and the partial arc lengths (13) of the driving gear 4a subject to the rolling condition have the same length. However, the toothed rolling curve radii $r_1(\varphi_1)$ and $r_2(\varphi_2)$ depending on the angular positions φ_1 and φ_2 are selected in such a way, that the result is a transmission behavior according to equation (4). With a constant center distance (28a) of the noncircular gear wheels (3a), (4a) the generally valid rolling curve function is given in polar coordinates by

$$r_1(\varphi_1) = \frac{a}{i+1} = \frac{a}{i_m \cos\varphi_2 + 1} = \frac{a}{\sqrt{i_m^2 - \varphi_1^2} + 1} \qquad (5) \quad \text{and}$$

$$r_2(\varphi_2) = \frac{a\,i}{i+1} = \frac{a\,i_m \cos\varphi_2}{1 + i_m \cos\varphi_2} \qquad (6)$$

The desired transmission function $i(\varphi)$ is enforced with the illustrated positional arrangement of the sprocket (10) relative to the driven gear wheel (3) by the noncircular gear wheel pair (3a), (4a) in such a way, that the angular velocities ω_2 vary between a

minimum: $\omega_{2\min} = \frac{\omega_1}{i_m}$ at $\varphi_2 = 0$ and $h = r_0$

maximum: $\omega_{2\max} = \frac{\omega_1}{i_m \cos\varphi_{2\max}}$ at $-\varphi_{2\max} = \varphi_2 = +\varphi_{2\max}$ and $h = r_0 \cos\varphi_{2\max}$

resulting in a constant chain velocity at each position φ_2.

Those skilled in the art will recognize that equations 5 and 6 define mathematical functions known as cardioids which is a closed curve between 0° and 360° resembling the shape of a heart. More specifically, the shape of rolling curve section 9 in the preferred embodiment is generated as a segment of the functions described by the polar equations (5) and (6). In the preferred embodiment, rolling curve sections are formed as that segment of a cardioid which most closely resembles a circle. The cardioid is preferred because it is mathematically correct. In this connection it is to be noted that Figure 2 of the drawings is schematically illustrating the radii, r_1, r_2, of rolling curve sections 9 for drawing clarity purposes only.

While a cardioid is preferred, the advantages of the invention may still be realized (to a lesser extent) with rolling curve sections 9 of a different configuration. That is, sinusoidal or circular configurations for example, can be shaped to meet the requirements of a maximum radial distance at the polygon corner and a minimum radial distance at the midpoint of the polygon side line. It should also be recognized that "pitch circle" when used for defining the rolling curve sections (which carry the spur gear teeth) is not technically correct because a "circle" is not present. "Pitch circle" is used because it is a well known term in gearing literature describing gear teeth. "Rolling curve" is also well known in the gearing literature and is used herein in its general conventional sense.

Noncircular gears can be economically manufactured today for complicated rolling curve shapes. In addition they can be realized just as simply for the frequent case of round link chain sprockets (10) with unequal pitch angles as with equal pitch angles.

Feasible gear ratios at the equivalent polygon center of the corners or corner middle, i_m, are calculated for

equal pitch angles

$$i_m = \frac{\varphi_1}{\sin \varphi_{2\max}}$$

unequal pitch angles

$$i_m = \frac{\beta_1 + \gamma_1}{\sin \beta_2 + \sin \gamma_2}$$

and for a given number of teeth c of the chain sprocket the equations to calculate the various angles are given by

$$2\alpha_2 = 2\pi / c \qquad \alpha_2 = \beta_2 + \gamma_2 \quad \text{and} \quad \beta_2 = \mathrm{a}\tan \frac{\sin \alpha_2}{\frac{t-d}{t+d} + \cos \alpha_2}$$

In case of round link chains (8) for reasons of symmetry with unequal pitch (t – d) and (t + d) only an even number of arc sections, *e*, can be realized at the driving noncircular gear wheel with typical parameters for round steel chains such as $(t-d)/(t+d)=0.5$ in the following table gear ratios for the example of a chain wheel with six corners with $\varphi_{2\max} = 30°$ are calculated with $c = 6$ follows $\alpha_2 = 30°$ and $\beta_2 = 20.1°$ and $\sin\beta_2 + \sin\gamma_2 = 0.515583$.

number of arcs "e"of drive gear wheel	round link chains unequal pitch angle		average gear ratio	roller chains equal pitch angle		average gear ratio
(13)	$\beta_1+\gamma_1$	$i_m=1.94(\beta_1+\gamma_1)$	$i_{aL}=2c/e$	φ_1	$i_m=2\varphi_1$	$i_{ar}=c/e$
1	-	-		π	6.283	6.00
2	π	6.093	6.0	$\pi/2$	3.142	3.00
3	-	-	-	$\pi/3$	2.094	2.00
4	$\pi/2$	3.047	3	$\pi/4$	1.571	1.50
5	-	-	-	$\pi/5$	1.257	1.20
6	$\pi/3$	2.031	2	$\pi/6$	1.047	1.00
7	-	-	-	$\pi/7$	0.897	0.857
8	$\pi/4$	1.523	1.5	$\pi/8$	0.785	0.750
...	...	...	...	...	...	...

In most cases gear ratios between 1.5 and 3 will be sufficient resulting in no limitations to applications.

Figures 2 and 3 illustrate the embodiment of noncircular gear wheels with such gear ratio adjustment consisting of a noncircular driven gear wheel (3a) and a noncircular driving gear wheel (4a), where the driving gear wheel (4a) is positioned to the driven gear wheel (3a). In such an arrangement, the respective smallest angular velocity coincides with the corners (29a) of the chain wheel-polygon (29) and the respective increased velocity occurs at the middle of a polygonal straight line (29b). In case of round link chains (8) the pitch curve radius (13a) of the driven gear wheel (3) is greater in the middle of the shorter equivalent polygon straight line (30) than in the middle of the longer equivalent polygon straight line (31). The spur gear (2) may have one or several noncircular gear meshes (11), where at least the last mesh (12) has to be embodied as noncircular gear mesh (14).

In case of a pivot chain (7) the driven gear wheel (3a) has at the pitch curve circumference (13a) a number of continuous rolling curve sections (9b) which is equal to the number of corners of the chain wheel (10). Each of these rolling curves (9a) forms an arc "b".

Furthermore in case of a round link chain (8) the driven gear wheel (3a) has at the pitch curve circumference (13a) a number of continuous rolling curve sections (9a), which is twice the number of teeth c of the chain wheel (10).

The drive gear wheel (4a) is also furnished with such continuous rolling curve sections (9b) at the pitch curve circumference (13).

In the case of the pivot chain (7) the drive gear wheel (4a) has an arbitrary number of rolling curve sections (9b) equal to or more than one. In the case of round steel chains (8) the drive gear wheel (4a) has an even number of continuous rolling curve sections (9b).

Thus, the number of continuous rolling curve sections (9b) on the drive gear (4a) corresponding to the pitch angle (15) is adjusted to the choice of the gear ratio to the driven gear wheel (3a). The geometric shape of the continuous rolling curve sections (9) is embodied in such a way that at a constant angular drive velocity ω_1 the driven angular velocity ω_2 follows from $(\omega_2 = \omega_1 / i)$ by multiplying the gear ratio at the corner middle with the cosine of the driven angle φ_2, which results in $i = i_m \cos\varphi_2$.

The continuous rolling curve sections (9) are of such geometry, that the gear ratio "i" can be approximated by basic or composite polynomials, trigonometric functions, Fourier series, sections of eccentric circular arcs, or periodic or mathematical approximating functions.

Fig. 4 illustrates the kinematic relations at sprocket (10) with the notations used. Herefrom follows velocity v_1 and velocity v in horizontal direction. The lever arm size h is thus a function of the driven or rotational angle φ_2 at the driven angular velocity ω_2.

Fig. 5 illustrates rolling curve sections (9) of the driven gear wheel (3) concave unilaterally bent transition arcs (16) at the point of intersection (17) touching the rolling curve sections (9) at tangential points. Instead of tangential transition arcs (16) at the rolling curves (9a) doublesidedly bent transition curves (18) or undulating curves can also lie within the tangential touching points (19) of the continuous rolling curve sections (9).

The adjustment curves (18) are symmetrical and can be described mathematically at least by a polynomial of fourth order or a modified trigonometric function being at least of the form $x \sin x$.

At the angular position of the intersection point (17) of the continuous rolling curve sections (9) the adjustment curve (18) and the transition arcs (16) have a radius of curvature equal or greater than the radius of a manufacturing tool (20).

According to Fig. 6 the driven gear wheel (3) is manufactured in at least two pieces intersecting at points (17). A primary part (21) can be assembled with a secondary part (22) in such a way, that concave sharp intersections of the rolling curves (23) are created without transition arcs (16) or adjustment curves (18).

Fig. 6B – 6D illustrate, that every second rolling curve section (9) is absent and an arc gap (24) is reduced radially down to a centering radius (25). The arc gap (24) can be used both as a tool recess and as a centering means for the respective complementary part. It should be noted that if the rolling curve sections (9) are even numbered, the primary part (21) and secondary part (22) are identical in the preferred embodiment. This results because the centering radius (25) forms a hub which is about one-half the thickness of the pie shaped sections forming the rolling curve sections (9) at their circumference.

A possibility exists for the practical case, if the polygon effect cannot completely be compensated by transition arcs (16) or adjustment curves (18), to provide a compensation with an additional or intermediate noncircular toothed driven gear wheel (3a) or preferably, an additional noncircular toothed driven gear wheel (4a) and driving gear wheel (3a) (with driving gear driven by the driven noncircular gear of the first gear arrangement to produce a cascaded gear set) is used. In all cases the additional gears must be correctly located circumferentially on their rotating axis relative to the transition arcs (16) or adjustment curves (18) for compensation.

Further details result from the list of reference symbols in connection with the drawing.

List of reference symbols

1	chain wheel axis
2	spur gear
3	driven gear wheel
3a	noncircular toothed driven gear wheel
4	driving gear wheel
4a	noncircular toothed driving gear wheel
5	drive side
6	driven side
7	steel pivot chain
8	round link chain
9	continuous rolling curve section
9a	rolling curve on driven gear
9b	rolling curve on driving gear
10	chain wheel (sprocket)
11	noncircular gear wheel mesh
12	last gear mesh
13	pitch circle circumference
13a	pitch circle radius
14	noncircular gearing
15	pitch angle
16	transition arc
17	point of intersection
18	adjustment curve
19	tangential touching points
20	manufacturing tool
21	primary part
22	secondary part

23	intersection of rolling curves
24	arc gap
25	centering radius
26	equivalent polygon straight line
27	partial arc length of drive gear 3a
28	center distance "a"
29	polygon
29a	polygon corners
29b	polygon straight line
30	shorter equivalent polygon straight line
31	longer equivalent polygon straight line
a	center distance shown by reference number 28
b	arc of rolling curve section 9
c	number of teeth
d	thickness (diameter) of round link chain
d_0	chain wheel diameter
e	number of arc sections of driven gear wheel
h	lever arm
i	gear ratio
i_m	gear ratio at the corner midpoint of the polygon
i_{aL}	average gear ratio of round link chains
i_{ar}	average gear ratio of roller chains
r_0	chain wheel radius
r_1	rolling curve radius of driving gearing
r_2	rolling curve radius of driven gearing
t	pitch

v longitudinal chain velocity

x horizontal coordinate

ω_1 driving angular velocity

ω_2 driven angular velocity

φ_1 driving angle/angle of rotation

φ_2 driven angle/angle of rotation

γ_1 pitch angle

γ_2 pitch angle

β_1 pitch angle

β_2 pitch angle

α_2 chain wheel pitch angle

$2\alpha_2 = 2\pi/c$ pitch angle

The invention has been described with reference to a preferred embodiment. Obviously, alterations and modifications will suggest themselves to those skilled in the art upon reading and understanding the Detailed Description of the Invention set forth herein. For example, the specific embodiments of Figures 2 and 3 show a noncircular driving gear in toothed contact with a noncircular driven gear. Obviously, an intermediate noncircular gear can be inserted between the driving gear 3a and driven gear 4a. The gear ratios between noncircular driving and driven gears can be varied within the ranges discussed above, but circular gears (3, 4) as shown in Figure 1 can be employed with the noncircular gears to produce any desired gear ratio. The embodiments have been discussed with reference to steel chains. Other chain compositions such as thermoplastic chains can be employed. Also, those skilled in the art will recognize that "driving" and "driven" is used in the context of two gear wheels in drive relationship with one another. Thus a sprocket or pinion is a driving gear wheel driving a "driven" gear wheel. The "driven" gear wheel is driving a chain wheel and it that sense is a "driving" gear wheel. It is intended to cover all such modifications and alterations insofar as they come within the scope of the present invention.